1

Introduction

1.1 Gas Lubrication in Perspective

Gas lubrication is a special branch of fluid film lubrication distinguished by the fact that the lubricating fluid is a gas that is a *compressible* fluid. In fluid film lubrication, the friction between two solid surfaces is reduced through the introduction of a fluid under pressure that flows between them. The pressure flow of the fluid may be induced by shearing the wedge-formed film, caused by the relative motion of the solid surfaces, in which case the lubrication is known as *hydro- (aero-) dynamic*, and the bearing as self-acting. When the flow, on the other hand, is induced by external pressurisation, the lubrication is known as *hydro- (aero-) static*, and the bearing as externally pressurised.

The thickness of the gas film that separates the bearing surfaces may range from sub-micrometre to several tens of micrometres depending on the size of the system and the application at hand. The bearing surfaces might be rigid or flexible, permeable or impermeable, flat or shaped, smooth or grooved. In short, a large variety of sizes, configurations and applications are available, which makes the topic both wide and rich in possibilities.

It would be difficult to imagine today's technology, from the aluminum foil in the kitchen through surgical tools in the health service to the computer disc drives, without gas bearings. Yet gas lubrication remains in relative obscurity when compared with other branches of lubrication: in the mind even of the engineer, the word "bearing" is synonymous with "rolling-element bearing". Although gas bearings are nearly as old as liquid film bearings (which are older than rolling-element bearings), they are still looked upon as a *novelty*. In the following sections, we shall try to situate gas lubrication from a historical point of view, in terms of their utility, from the morphological side, and in regard to their past and present application domains.

1.1.1 Short History

The evolution of gas lubrication cannot be isolated from that of fluid film lubrication in general. However, in the earlier stages, interest in gas lubrication was only marginal; fluid film lubrication being considered mainly *liquid* film lubrication, and the possibility of using a gas looked upon with some suspicion. (Yet, it is ironic that one of the earliest lubrication experiments pointed out that air was acting as a lubricant.) The following historical outline will therefore deal with fluid film lubrication in general while focusing mainly on gas lubrication. It is, further, based on the references (Dowson 1979; Fuller 1990; Gross 1962; Pan 1990) in increasing degree of importance.

Three periods may be identified in the evolution of gas lubrication.

The first is the period of *pioneering*. Evidence of *aerodynamic* lubrication was made known by G. A. Hirn in 1854. He conducted experiments on a "friction balance" (which is a half journal bearing with a means of measuring the

Air Bearings: Theory, Design and Applications, First Edition. Farid Al-Bender.

Companion website: www.wiley.com/go/AlBender/AirBearings

torque) to show that, with a sufficiently high speed of rotation, "non-viscous fluids such as water and air" may be dragged into the bearing; when the speed was reduced to a certain value, the friction force became at once enormous. This is very remarkable since it was one of the earliest pieces of evidence of *fluid* film lubrication.

It was N. Petrov, in 1883, who made the first significant attempt to analyse *theoretically* the friction effect of film lubrication, recognising that it was the film, rather than the bearing materials, that was of prime consideration. However, his unrealistic assumption of *uniform* film thickness, made his results rather limited in importance; falling short of establishing a lubrication theory. He was able, however, to derive the viscous-friction law that bears his name today.

In that same year 1883, B. Tower reported very important experimental results, showing the pressure distribution in a partial journal bearing, and thus pointing out that *pure* fluid film lubrication, that is a fluid film that is able to separate the two relatively sliding surfaces entirely, was possible.

Three years later, in 1886, Osborne Reynolds, unaware of Petrov's theory, was able to explain the results of Tower. He successfully derived the basic differential equation that describes the mechanism of fluid film lubrication that today bears his name. He published at the same time certain solutions that agree well with the results of Tower. Reynolds, however, made no reference to the experiments of Hirn, nor indeed considered the possibility of gas lubrication.

This type of lubrication was not discussed again until A. Kingsbury, unaware of Reynolds' theory, built an air lubricated journal bearing in 1886. (This was based on his earlier rediscovery of air lubrication in a compression piston device of his making.) Having been made aware of Reynolds' theory, he was able to obtain excellent pressure distribution data and publish them in 1887. This was perhaps the first time that air lubrication was demonstrated without any doubts.

In 1913, W. J. Harrison returned to the subject of gas lubrication with a renewed zeal. For the first time, the *pressure dependence of the density* was formally addressed. On the basis of simplified consideration, he introduced the concept of *isothermal state* whereby the density is proportional to the pressure; an assumption that to this day is widely accepted. He also pioneered the idea of numerical computation, laboriously producing solution to the air bearing problem that compared very well with the experimental data of Kinsbury. He characterised the effect of compressibility in the parameter Λ, known in his honour as the Harrison number.

At this juncture, a word must be mentioned about *hydrostatic* lubrication. As the virtues of fluid film lubrication generated by motion of the bearing surfaces became recognised, attention was naturally diverted to the limitations imposed by excessive loading or low surface speeds. Lord Rayleigh was the first to present an analysis of a hydrostatic thrust bearing, in 1917. He derived expressions for the logarithmic radial decay of pressure from a central supply in the space between parallel circular discs, the resisting torque and the limiting load capacity of the bearing. He pointed out the practical importance of situations that could be accommodated by such bearings, emphasising that with the proper geometrical accuracy of the bearing surfaces and the cleanliness of the lubricant, there should be *no wear* of the solid surfaces, which should never come into contact. (The principle and application of hydrostatic lubrication had already been demonstrated, and patented, by L. D. Girard in 1865.)

Until this time, industrial applications of gas lubrication was hardly in evidence, with the notable exception of high speed spindles for the textile machinery for which patents were awarded in 1906 and 1909. Also, the advantage of high speed grinding, with the aid of air bearings, was recognised in 1909. Otherwise, research in fluid film lubrication concentrated mostly on *viscous* liquid bearings until the end of World War II.

The second period, termed by Pan (1990) as *the Golden Era*, began in the early fifties. Already, in the mid-forties, gas lubrication had played a key role in the preparation of high grade nuclear fuel, focusing attention once more on the possibilities of that technology. Hence, serious concerted effort, including of course material investment, was initiated to develop the potential of gas lubrication and to promote it as a technology base to advance the art of mechanical engineering. Sufficient motivations and numerous possible applications (that will be outlined in the next sections) existed to justify this.

Research groups were formed and programmes were initiated both in the USA and in Europe. Regular meetings, conferences and symposia were held and the information, in the form of minutes, reports or proceedings were disseminated to the involved researchers. The driving force behind this activity came from the areas of: closed-loop process equipment, inertial sensors, tape transports, disc drives and aircraft auxiliary machines.

The first International Symposium on Gas Lubricated Bearings was sponsored by the US Office of Naval Research in 1958, and convened in Washington D. C. The proceedings presented many significant advances in gas lubrication. Later, the University of Southampton (UK) convened regular symposia that attracted many participants from East and West Europe. Initially, attention was focused mainly on commercially viable applications. Ties with the US groups were established and joint conferences took place. This three-way dialogue continued many years until the latter part of the seventies, and was the driving force behind hundreds of scientific papers and publications. This period witnessed also the emergence of the first *comprehensive* text books on gas lubrication by: Gross (1962), Constantinesu (1969) (1962, in Rumanian, and 1969, English translation); Grassam and Powell (1964); MTI's lecture notes: Design of Gas Bearings (Wilcock 1969).

Hydrostatic (and aerostatic) lubrication was for the first time identified as a distinct mode of lubrication, and systematised by D. D. Fuller in 1947; hitherto, there had been only sporadic examples of externally pressurised bearing applications. The principle of external pressurisation was exploited in the field of gas bearings to an even greater extent than self-acting behaviour. They overcome low-speed operating problems and permit satisfactory stiffness while providing low friction, and hence small heat generation and minimal thermal distortion. Hybrid combinations, of self-action with external pressurisation began to come more to the fore, since self-acting gas bearings are otherwise limited in load capacity.

Finally, towards the latter part of the seventies, support for gas lubrication research began to wear thin. Although the accomplishments had been, thus far, quite impressive, there were no tangible signs of return on investment. For this to happen, considerable advances in wear control had to take place first, i.e. the gas lubrication research programme had "outraced itself" (Pan 1990). Thus, with the exception of air lubrication in magnetic storage systems, activities in gas lubrication research slowed down to a snail's pace.

This brings us to the third period of gas lubrication: *the present*. Gas lubrication is now recognised as a fully developed applied science; gas lubrication has become one of the proven principles available to innovators in their perpetual effort to contrive means for higher quality of living. Although gas lubrication research began to slow down considerably at the beginning of the eighties, the ever rising demands of modern technology have begun to breath new life into the subject. Requirements of high precision and high speed in machine tools and metrology, non-contact gas seals, computer storage heads, special test facilities, and many other yet unidentified applications, will continue to press for more advances and gas bearings will have a significant role to play for some time. Perhaps the main feature of the present era is the consolidation of the hitherto achieved results and the further promotion of gas lubrication in technology. "It is an immediate challenge that the practice of gas lubrication be incorporated into the regular mechanical engineering curriculum in our higher education institutions and professional level programs." (Pan 1990).

1.2 Capabilities and Limitations of Gas Lubrication

The capabilities of gas lubrication may be directly related to the physical properties of gases. A gas is a very *stable* lubricant; it does not vaporise, solidify, cavitate or decompose at extreme temperature ranges. Gas bearings can thus be used in turbo-compressors operating at 650 °C, or at the other extreme, for cryogenic units at temperatures nearing absolute zero. Furthermore, gas viscosity is little affected by temperature, as compared to a liquid, retaining satisfactory values over wide temperature ranges. Gases are comparatively free from the adverse effects of radioactivity and are tolerant to hazardous environments. Their low viscosity (which is both an advantage and a disadvantage) results in very low friction coefficients and hence low frictional losses and their associated temperature rise. Furthermore, as in other fluid-film lubrication cases, the absence of surface-to-surface contact leads to negligible wear and implies quasi-infinite bearing life. Gas bearings are particularly suited to precision instruments owing to the form-averaging effect of the fluid film, their low noise during motion and their zero friction in the absence of motion. Air, in particular, is abundantly available, comparatively clean, ecologically sound lubricant; self-acting air bearings such as computer flying-heads are automatically *submerged in air*.

The limitations of gas bearings are mainly due to the compressibility of gases. Thus, by industrial safety as well as by dynamic stability limits, externally pressurised gas bearings are limited in supply pressure to around 10 bar (in industrial environments, often no higher than 6 bar-gauge is available), as compared to hydrostatic bearings, which can be fed with up to a couple of hundred bar. Furthermore, the low viscosity of gases means a higher rate of gas consumption, in comparison to liquid bearings, for the same bearing gap height. This entails the use of very small bearing gaps which, in turn, generally demands a closer control over manufacturing tolerances, surface finish, thermal and elastic distortions, and alignment. As for aerodynamic bearings, unlike hydrodynamic bearings, whose load capacity (or mean pressure) increases proportionally with sliding speed, compressibility in self-acting gas bearings results in saturation of the build up of mean pressure as the sliding speed is increased. Thus, a mean pressure build-up of around twice atmospheric pressure is about the maximum one could reach in realistic situations. Secondly, the combination of compressibility and low viscosity results in gas bearings being *poorly damped*. Pneumatic hammer instability is a well known phenomenon in externally pressurised air bearings. One of its main consequences is to limit the maximum allowable supply pressure, and consequently the load capacity and stiffness. In self-acting bearings, on the other hand, one often encounters self-exited whirl instability.

1.3 When is the Use of Air Bearings Pertinent

In view of the above-mentioned features of gas lubrication, two generic application areas can be identified, namely, high-speed and high-precision applications. The first follows from the low viscosity of gases; the second from the film-averaging effect. A combination of the two is also an emerging industrial need answering demands for mastering, inspection, etc., of CDs, wafers and so on. Besides these, a wide range of particular applications are also known, e.g. in the nuclear industry, magnetic HDDs, etc. Presently, gas-lubricated bearings are found in hundreds of essential applications. These may be classified as follows:

- Machine construction:
  - ultra-high-speed grinding and drilling spindles
  - slideways of (NC) machine tools
  - index tables
  - gyroscopes
  - (miniature) gas turbines and turbo-compressors, etc.
- Precision and metrology:
  - reference spindles and rotary tables for roundness measurement
  - 3D coordinate measuring machines
  - telescope and antenna supports
  - laser plotters for printed circuit boards
  - micro-manipulators, etc.
- Data retrieval systems:
  - transport of magnetic tapes
  - magnetic read/write heads for computers
  - disc supports, etc.

Many other applications fall into these categories such as: zero-gravity test facilities, diamond turning machines for large optics, low-friction tension controllers, hanging robot arms, etc. Generally speaking, air bearings *can* also replace any bearing application that falls within their load capabilities.

1.3.1 What Hinders the Extensive Application of Air Bearings

It has been apparent that the most important drawbacks of gas bearings are their comparatively low load-carrying capacities, high gas consumption (in the case of externally pressurised bearings) and their poor damping. The understanding and partial overcoming of these problems belongs to the realm of design theory. In this respect, there are still a number of unsatisfactorily solved problems, in particular those related to dynamic behaviour, stability and added damping, not to mention other minor problems. Gas bearing research, although great in quantity and quality, remains to a large degree scattered.

On the other hand, we have the problem of manufacturing. If air bearing application is to be widely promoted, low cost air bearing components, of well defined characteristics and, preferably, in somewhat standardised form, have to be available on the market, so that they can be easily purchased and assembled for the particular application. Up to the present, only flat (and some restricted types of radial) aerostatic bearings are available in this form commercially. In most cases, an air bearing application has to begin at the design/drawing stage through the details of manufacture and assembly, thus making the cost too high to be commercially viable. Evidently, the problems of design and of manufacture are strongly interconnected: a clear, thorough and easy-to-use design theory and environment leads to relatively simplified manufacturing requirement, while the facility of manufacture can allow, in its turn, for better design.

The third aspect is the awareness of engineers of the *existence*, practicability and reliability of gas bearings. Air bearings are still treated as a curiosity rather than an established technology. Furthermore the tools to design them do not lie directly at the finger tips of the design engineer. There could be too many design parameters and performance characteristics to be managed, let alone optimised, in a design process.

The author believes that it is high time that these three aspects were seriously taken into consideration in order that a proven engineering art occupy its rightful place in modern technology. This book is a modest step in that direction.

1.4 Situation of the Present Work

1.4.1 Short Overview of Gas-bearing Research at KULeuven/PMA

Gas bearing research began at the KU Leuven, department of Mechanical Engineering, since the early 1970s. It concentrated initially on the development of design methods to improve aerostatic bearing characteristics, namely, to increase the load-carrying capacity and the stiffness while reducing the gas consumption.

One of the basic design ideas explored was the use of a *convergent* gap geometry. This was found to yield bearing characteristics that are far superior to those of the conventional uniform gap bearing. Since the pressure distribution in the bearing depended now not only on the inlet pressure but also on the relative *conicity* of the film, the load capacity was found to increase dramatically with decreasing nominal film thickness. This meant an increase in the stiffness with increasing load, on the one hand, and an increase in the load capacity for a given mass flow rate (or air consumption) for a given mean air gap on the other. Furthermore, the convergent shape of the bearing gap resulted in reduced sensitivity to surface irregularities. At the same time, methods of manufacturing the required bearing profile were suggested, such that the cost of fabrication was kept very low.

This design idea found also an important extension: the compliant gap bearing. By making the bearing surface compliant, e.g. a thin plate, clamped at the inner or the outer edge, the conicity was allowed to vary with the mean bearing pressure. Thus, with the proper choice of design parameters, the bearing could be made *infinitely* stiff over a wide range of loads. A design methodology was developed for this type of bearing, based on the solution of the thin-plate bending problem simultaneously with the bearing pressure distribution problem.

Beside these two major topics, a considerable amount of work was devoted to the investigation of the problem of pneumatic stability of aerostatic bearings and of systems that incorporate them. It so happens that inherently compensated, convergent gap bearings were also less prone to pneumatic instability than their conventional, orifice-compensated pocketed-bearing counterparts.

These and many other related topics resulted in a considerable number of important publications, which are reviewed, in the framework of the development discussed above, in (Snoeys and Al-Bender 1987), which overviews the many outstanding results of two doctoral theses written in the course of this research programme (Blondeel 1975; Plessers 1985). This research was further consolidated and extended, especially in regard to entrance flow, tilt characteristics, and dynamic behaviour, in the thesis (Al-Bender 1992). Furthermore, understanding the dynamic characteristics of the air film led to the development of various types of actively controlled and compensated bearing concepts enabling the realisation of virtually infinite stiff bearings (Al-Bender 2009).

Knowledge of the film flow in pressure-induced turbulent regime led to the design of externally pressurised, over-expansion *hanging* air bearings as a curiosity, which is not, however, without potential applications.

Another type of bearing that has been developed in that same period is that of externally pressurised foil bearings, which also might hold certain advantages regarding manufacturing cost and ease of application.

More recent work has focused on mainly self-acting air bearings for miniature (ultra-) high-speed applications, in particular meso and micro turbines. Both compliant-surface as rigid-surface, self-acting and externally pressurised gas bearings have been investigated in two doctoral theses (Vleugels 2009) and (Waumans 2009) in the framework of ultra-high-speed miniature turbines. This is on the one hand, on the other hand, two other doctoral works have treated ultra-high-precision bearings; one regarding active mechatronics positioning systems (Aguirre 2010), while the other treated the design of nanometre precision rotary tables (Cappa 2014). Recently, another thesis was completed treating tilting-pad journal bearings (Nabuurs 2020), while work is ongoing regadring methods and techniques to of adding damping to high-speed journal-bearing systems.

This book draws on the long and rich experience in gas lubrication at KU Leuven and elsewhere in the world, which forms the basis for compiling it. It is hoped that this book should form an appropriate medium for propagating this knowledge in the research and industry world. In particular, while reviewing salient research findings from the state of the art on the subject, the book endeavours, in the first place, to distil essential knowledge about the fundamental aspects in theory, design and practice of air bearings.

1.5 Classification of Air Bearings for Analysis Purposes

As has been mentioned earlier, there is a broad range of air-bearing configurations, morphologies and working principles, which might make it difficult to deal with in concise and systematic manner. Moreover, there are various ways to classify any collection of items, depending on the specific objectives, the analysis approach or the utility of the results. We shall adopt a classification that makes the build-up and analysis of air bearing systems easier to follow. This classification is also a generic one, being used in other fields of physics and engineering, namely:

- *Dynamics*: relates to the manner in which the bearing force is generated. Here, we have three possibilities:
  - Aerostatic or externally pressurised (EP): in which pressurised air is introduced into the gap from an external means, e.g. a compressor.
  - Aerodynamic or self-acting (SA): in which the pressure in the bearing gap is generated by relative motion of the bearing surface w.r.t. each other. This could be one or a combination of:
    * sliding or tangential motion
    * squeeze or normal motion.

  - Hybrid, where both the aerostatic and the aerodynamic effects are present by design. This category is, however, very specific and is mentioned here only for the sake of completeness.
- *Kinematics*: relates to the motion and morphology of the bearing element. Here, we have many possibilities:
  - Motion in a plane: flat bearings
  - Rotation around a line: cylindrical and conical bearings
  - Rotation around a point: spherical bearings.

  These different categories, excepting the squeeze-film type, are depicted in Figure 1.1.

  Parallel to this, it might be also useful to subdivide all the previous types into:
  - Bearings with rigid surfaces, including porous bearings, which constitute the majority of applications.
  - Bearings with compliant surfaces, which comprise:
    * Foil bearings of the bump and tension type, where the compliance is evenly distributed over the bearing surface. The mode of lubrication in that case is elasto-hydrodynamic lubrication (EHL) or, more specifically, elasto-aerodynamic lubrication (EAL), including of course the elasto-aerostatic case.
    * Bearings in which the surfaces are rigid but are flexibly supported to adapt the gap shape to the load dynamics. The most salient example of this type is the tilting-pad bearing (TPB).

    This subdivision is indicated in the last row of Figure 1.1.

  Each of these types of bearing can be further divided into:
  - Single boundary: resulting in bearing pads or shoes whether of the flat, cylindrical, conical or spherical types. These could be combined to achieve the functionality of the next category of bearing morphology.

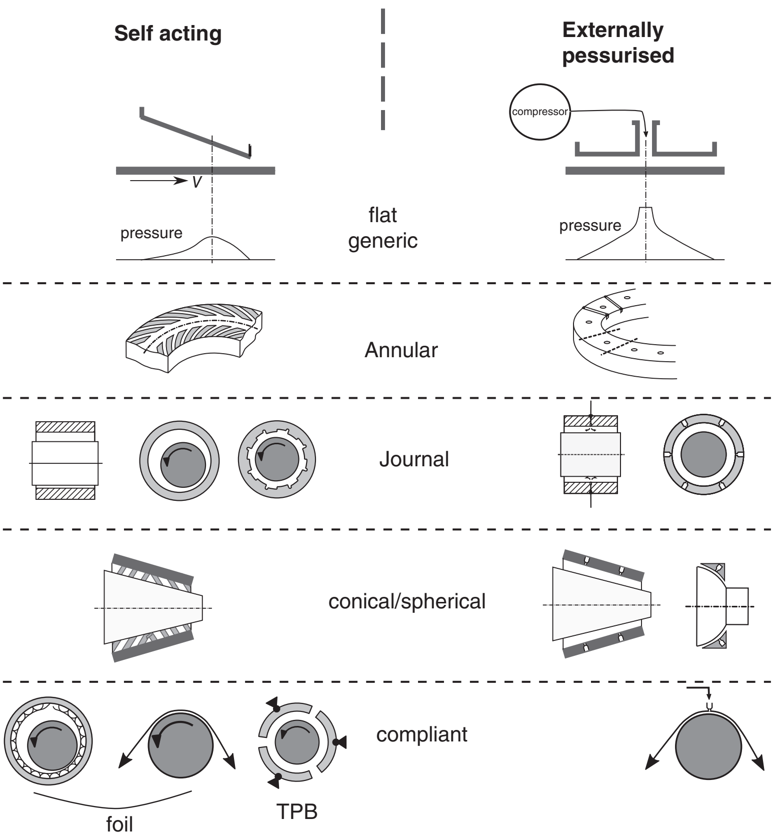

Figure 1.1 Classification of air bearings according to pressure generation (dynamics) and morphology (kinematics). The bottom row depicts compliant-surface bearings.

– Double boundary: resulting in (flat) annular bearings, journal bearings (of both cylindrical and conical type) and spherical ring bearings.

Let us note, however, that not all of these types and bearing configurations are treated in this book. Nevertheless, since specific problems (or situations) exist only as tangible (thus, in a way, unique) manifestations of general ones, the models proposed and the solutions obtained should enjoy some degree of general validity. In particular the results obtained for flat circular bearings should hold also true for *radial* bearings formed by bending the flat surface into a cylindrical one. They should hold at least qualitatively true for *square*- and *sector*-shaped bearing pads. Likewise, knowledge of annular thrust bearings and of journal bearings should put the reader in good stead to deal with conical or spherical bearings.

This book focuses primarily on the underlying theory, since it is that that can form the basis of a good design methodology; no amount of empirical data can replace a proven theory. Yet, theories must be verified by practice and thus particular attention is paid, in this work, to experimental verification. Comparison is carried out with experimental results taken from various sources in the open literature as well as experiments carried out in the laboratories of KU Leuven.

1.6 Structure of the Book

The book has basically five constituent parts:

1. Fundamentals: general modelling, feed flow, Reynolds equation, solution methods, thermal aspects (placed at the end of the book).
2. Flat circular aerostatic bearings: radial flow, basic bearing characteristics, dynamic behaviour and stability.
3. Aerodynamic and hybrid bearings: aerodynamic action, thrust bearings, journal bearings, whirl stability.
4. Other bearing types: tapered-pad bearings, porous bearings, foil bearings, hanging bearings.
5. Active bearings: servo-controlled bearings, squeeze-film bearings, case study: linear slide.

This sequence might be relevant from a scholastic point of view. The advanced reader might prefer to explore the book selectively. There might be some repetition here and there, being dictated by the self-sufficiency of individual chapters or by these coming from contributing authors.

1.6.1 Chapter per Chapter Overview

Chapter 2 formulates the problem, starting from the bearing configurations and the flow-describing equations. Simplifying assumptions are imposed on both of the former in order to facilitate the derivation of practicable working models. Two models are proposed: a refined model for the solution of the feed-and-entrance flow problem in the quasi-steady case, and the conventional Reynolds model with a feed-flow restrictor. Normalisation of the problem is also considered in order to identify the basic dimensionless parameters influencing the behaviour and to systematise subsequent treatment; similitude criteria are deduced. Methods of solution are outlined and discussed.

Chapter 3 proposes methods of solution to the *channel entrance* problem, i.e. the problem of flow development slightly upstream and downstream of the bearing gap entrance. Two situations are treated: (i) pressure-induced flow, such as that obtaining at the entrance of an aerostatic bearing and (ii) shear-induced flow, such as that obtaining at the entrance of an aerodynamic bearing. Also a combination of the two situations is considered. A review of the relevant literature for each case is first carried out from which a method is selected, developed and established as a simple and reliable method of solution for a variety of thin-passage flow configurations. Although nowadays it might be a relatively simple matter to determine the flow field by a purely numerical, e.g.

the CFD method, it is nevertheless rewarding to derive such lumped-parameter methods, as carried out in this chapter. These methods afford us more straightforward means to establish the pressure boundary conditions required in solving the Reynolds equation, which characterises the major part of the bearing film. In that way, not only very fast solution algorithms can be devised to solve the bearing problem, but also useful insights into the design problem are gained.

Chapter 4 is dedicated to the Reynolds equation. First, we give a brief though complete derivation of that equation in its generic form, starting from the viscous laminar flow equations. Thereafter, with the aid of a coordinate transformation scheme, the equation is stated in most coordinate systems pertaining to diverse bearing configurations. Moreover, we consider interpretation of the kinematics for special cases pertaining to situations comprising uneven sliding surfaces having periodic surface features.

Neglected flow effects are categorised as boundary-condition related, mainly slip at the walls, and Reynolds-number related, comprising mainly inertia effects and, more particularly, turbulence. These are reviewed and some appropriate ”modified” Reynolds equations have been presented in order to deal with them. Inertia effects are considered at length together with derivation of the momentum and energy integral methods for a flow solution, leading to another type of modified Reynolds equations consisting in a system of simultaneous ordinary differential equations.

Chapter 5 implements the first method developed in Chapter 3, i.e. that pertaining to pressure-fed flow, to provide solutions to the problem of radial channel flow, relevant to circular, centrally fed (CCF) aerostatic bearings. The general trends of the flow are investigated and many comparisons are made of the pressure distributions, obtained by this method, with experimental data from various sources. Finally, owning to the good agreement of the results, the method is used to provide data for the correction coefficient of the restrictor model. This refined restrictor model is used, together with Reynolds model, for the treatment of the remaining problems (Chapters 6 and 7.)

Chapter 6 provides a systematic treatment of the basic characteristics of CCF aerostatic bearings under three categories: (i) static axial characteristics, namely, the load capacity, stiffness and air consumption as a function of the bearing design parameters (the dynamic characteristics are treated in Chapter 7); (ii) the problem of small amplitude tilt. The method of small perturbations is utilised to linearise the problem, which appears sufficient for obtaining fairly accurate results. Both static and dynamic tilt are considered in order to obtain a comprehensive idea about the problem. Experimental verification of the theoretical results is also reported; (iii) the effect of sliding on the otherwise *aerostatic* bearing. The influence of sliding on the load capacity, the stiffness, the flow rate, the centre of force, and the friction force are calculated, presented and discussed. In all these parts, the effect of the conicity, i.e. the shallow pocket around the feed hole, is particularly considered. All these aspects provide a basic understanding of the design problem and furnish a toolbox, as it were, for dealing with the issue of correct bearing design and optimisation

Chapter 7 is devoted to the study of the dynamic behaviour of circular, centrally fed aerostatic bearings and the problem of pneumatic-hammer instability. The various aspects of the problem are examined in conjunction with a literature survey of past treatments. Thereafter, a linearised model, based on the Reynolds/restrictor model is proposed, given certain constraints on the design parameters and the amplitude of oscillation, and its range of validity determined. Such a model is found sufficient to characterise the problem of small amplitude axial oscillations over a wide range of design interest. This methodology and model can be easily applied to other bearing configurations. For a given set of bearing design parameters, the model predicts the dynamic stiffness and damping as a function of the perturbation frequency. A simple equivalent system of springs and dampers (of positive or negative coefficients) is constructed to mimic the air film, which can be used as a design artefact. Stability criteria are deduced, and results are compared with experimental data.

Chapter 8 establishes the fundamentals of the aerodynamic action. Whereas in EP bearings the fluid pressure is provided from an external source, in self-acting bearings, the film wedge combined with viscous shear through the tangential velocity is the mechanism by which fluid is pumped into the gap in order to generate pressure;

this is the principle of the viscous pump, whose mechanical efficiency is shown to be very small. The qualitative difference between incompressible and compressible cases is examined at length, to clarify the limited load capacity of self-acting bearings. Thereafter, a number of generic aerodynamic bearing pads are reviewed to show basic properties, namely, the load capacity, static stiffness and damping, and the friction power dissipated. We consider nominally flat rectangular bearings to demonstrate theory, which could be extended to other situations. A considerable part is devoted to the derivation of the narrow-groove theory (NGT) of herringbone groove bearings and to the discussion of its basic properties. Damping in self-acting bearings can also become negative. This is so especially in spiral and herringbone groove bearings, and we demonstrate this by examining a bearing case.

Chapter 9 treats journal bearings (JBs) and their basic characteristics. We have considered two categories of bearings, namely, self-acting and externally pressurised bearings. The first category is represented by two types: plain and spiral grooved, which significantly differ in their static and dynamic behaviour. The second one comprises various variants of externally pressurised bearings, which are essentially used for low-speed precision applications. As such, they are characterised by high stiffness and, depending on the design, also high damping. Such bearings are also often used for high-speed applications, where they are sometimes referred to as hybrid journal bearings. This type has certain important advantages over the self-acting types, particularly in regard to high stiffness and dynamic stability. When ultra-high speeds are envisaged, it is generally not possible, from this analysis, to select a particular bearing type: the process entails optimisation to avoid the occurrence of whirl instability. In this chapter, we introduce the phenomenon of half-speed whirl and the associated critical mass above which the system will become dynamically unstable. This problem is more systematically treated in Chapter 10.

Chapter 10 treats in a fundamental way the dynamic whirl behaviour of gas lubricated journal bearing systems, which might be an important limiting factor on the performance of these bearing systems. Particular attention is paid to the study of the self-excited whirl phenomenon, which is often destructive, as well as to its remedies. First, the various kinds of rotor whirl are classified into synchronous whirl, being due to imbalance, and self-excited whirl, which sets in at high values of the rotational speed. Hereafter, the cross-coupled stiffness is identified as the root cause or the driving force of self-excited whirling. A simple but effective stability criterion is formulated that relates the maximum allowable cross-coupled stiffness to the other dynamic film parameters. Based on this stability criterion, some stabilising techniques from literature are assessed. Three methods to overcome (or at least postpone the occurrence of) self-excited whirl are proposed. In a first method, the design rules for achieving optimal stability with plain aerostatic bearings are outlined in a dimensionless way. The second stabilising technique tries to eliminate the driving force of self-excited whirl, namely by reducing the cross-coupled stiffness. This approach, in addition to being limited in effect, is not easy to implement in practice. Finally, a third and last strategy works by compensating for the destabilising effects from without the gas film; that is by tuning the dynamic characteristics of the bearing's damped support. This method is most popularly implemented in the form of o-ring supported bearing bush.

Chapter 11 treats foil bearings, which are obtained by making the bearing surface compliant rather than rigid, mostly by disposing a membrane around a shaft. After a review of the state of the art, foil bearings are classified in two major types, namely, (i) *tension* foils, where the surface is wrapped around a shaft by tensioning it, and (ii) *bump* foils, where the foil is not tensioned but rests on a bed of springs. Both cases are treated, although the largest part of the chapter is dedicated to idealised bump foils. Although these are quite complicated systems, it is possible to simulate certain aspects with a modified Reynolds equation where the gap height depends not only on the eccentricity but also on the local pressure via the dimensionless compliance parameter.

The steady-state characteristics, such as the load capacity and stiffness, are treated first, which, for this type of bearing depend not only on the bearing number and eccentricity ratio but also on the bearing surface compliance.

The dynamic characteristics, treated thereafter, are also highly influenced by the surface compliance, as that can be considered as a spring-damper system in series with the air gap stiffness. Furthermore, a two-dimensional rotor-dynamics whirl-stability analysis is carried out and some methods to improve the stability are discussed.

Chapter 12 treats porous bearings, which offer another possibility of introducing the (gas) lubricant into the bearing film, namely through an extremely large number of feeding holes that are of exceedingly small diameters, i.e. the pores. In this way, one achieves a distributed flow all over the bearing surface, which has certain important advantages, but also some drawbacks. This chapter deals with the static and dynamic fluid film modelling of porous bearings. The flow in the porous medium is assumed to be viscous and is governed by Darcy's law. A modified Reynolds equation, describing the flow in the lubricant film, is derived, taking into account the effect of slip flow, and a numerical solution process is developed and explained. First, the static characteristics of porous bearings are presented and discussed. In a second part, the dynamic bearing characteristics are defined and the perturbation method is applied to plain and cylindrical porous bearing configurations in order to obtain them. An expression for the dynamic film coefficients is given and their typical behaviour is discussed. The porous model is also validated with experimental and theoretical data found in the open literature, where good to excellent agreement is found for the static solution over a wide range of bearing parameters.

Chapter 13 treats the theory and practice of tilting-pad air bearings (TPB), in particular of the journal type. This bearing type is famed for its better dynamic, i.e. whirl stability as compared to other-rotary bearing principles. In order to build up the theory, we first look at the plane, infinitely long inclined-pad bearing case in order to establish its basic load, stiffness and damping characteristics. In a TPB, the inclined pad is supported by a pivot so that it can freely adjust its attitude. This requirement introduces additional dynamical aspects in the working of the pad, which need to be carefully formulated and dealt with in order to develop an effective design theory and practice. The greatest part of this chapter is then devoted to the case of journal TPB, where first the static characteristics are addressed followed by the whirl dynamics considerations where a methodology is developed to optimise the design of TPBs, pushing the whirl-onset speed as far as possible beyond the working range of the system. Finally, some fabrication aspects of these bearings are overviewed and discussed, since they are not as evident as for other bearing types.

Chapter 14 looks first at the different possibilities of achieving a bearing that *hangs* from a ceiling rather than *presses* against a floor, as is usual in most applications. It first situates the problem of hanging bearings, showing the advantages and possibilities that they can offer. Thereafter, it presents and evaluates the three possibilities of achieving a hanging bearing, namely, (i) usual thrust bearing combined with a magnet that attracts it against a magnetic-material ceiling, (ii) a vacuum or suction bearing, and (iii) the Bernoulli/over-expansion bearing where the gas flows in a turbulent, supersonic manner through the largest portion of the gap. This latter type has a certain potential as compared to the other realisation possibilities. Moreover, the theory of this type of bearing furnishes a good case study of turbulent-flow lubricating films. A reliable working model is thus derived, and experimentally validated, for use as design and optimisation tool. This new type of developed hanging bearing is very simple and effective, but requires large flow rates. It is easy to implement, requiring relatively large gap heights and thus not especially prepared ceiling surfaces. Suggested applications are: for hanging robot arms in micro-assembly cells; for handling delicate objects such as magnetic discs, chocolate, etc.; flat-wall climbing devices such as window-cleaning robots; or simply as elements to preload other thrust bearings, when light weight is required. The chapter makes also a pros–cons comparison between the three types of realisations mentioned above.

Chapter 15 firstly situates the problem of active compensation or servo control of gas films within the state of the art of air bearings. We see that there are various ways to influence and control the dynamic force in the bearing film, which can be narrowed down to two basic ways: (i) inlet/outlet pressure or flow control, and (ii) film geometry control; the latter having generally a broader frequency range than the former. Then it formulates the general film-force control framework and discusses it in connection with examples pertaining to thrust and journal bearings. Two other special cases are presented. Firstly, control of the traction force on the bearing surfaces and its application in wafer in-plane positioning is presented together with two solutions belonging to inlet/outlet pressure control and to film-geometry control respectively. Secondly, load generation by means of squeeze-film action presents a way of constructing bearings that do not require a pressure supply. The basic theory pertaining to this type of active bearing is over-viewed. From all of this, we see that active gas bearings represent an interesting extension to conventional, passive bearings, all the more so with the increasing utility of the mechatronics engineering methodology.

Chapter 16 describes the development of a linear slide with sub-micrometre accuracy requirement in six degrees of freedom, being chosen as a plausible industrial application for active air bearings, of which a first prototype has been developed. The first challenge is to optimise the design of the active air bearings, considering the coupled interaction between the air–film dynamics, structural flexibility, piezoelectricity and control, for which a multiphysics simulation model is developed and validated as tool for this task. The next step is the design of the active slide, analysing the interaction of several active bearings and their integration into a more complex system, where other effects, such as manufacturing limitations or calibration can determine the design requirements. The basic functionality of the active slide prototype is demonstrated. The implementation of calibration strategies and advanced control techniques should allow for the achievement of sub-micrometer accuracy over very large strokes.

Chapter 17 deals with the thermal aspects that arise during the implementation of air bearings both (i) in high-speed applications, where frictional heat dissipation can be an issue, and (ii) in precision applications, where thermal uniformity might be an issue. Regarding the first issue, we show by means of simple solutions of the thermodynamic energy equation that the heat generated by viscous friction, which may be quite appreciable, can hardly be evacuated by the air itself so that conduction through the bearing surfaces will be the only effective mechanism to accomplish this. It might suffice in certain situations to cool only one bearing member, e.g. the housing, which might simplify the task greatly. Formulas have been provided to quantify the various aspects of the thermal problem based on dimensionless parameters. Regarding the second issue, attention is focused on the thermal behaviour of the flow of centrally fed aerostatic bearings. A method of solving the energy equation simultaneously with the momentum equation is presented. It is based on approximating the convective terms in the energy equation, and using the mean density and viscosity values across the gap, all of which being considered sufficient for obtaining a quantitative idea about the thermal influences on the flow. The results show that the flow becomes isothermal in the viscous region regardless of whether the walls are thermally conductive or not. If the walls are isothermal, as is usually the case in gas bearing surfaces, the mean gas temperature will approach the wall temperature (assumed equal to the stagnation temperature) in the viscous region. For this case, typical behaviour of the polytropic exponent, in relation to the entrance reduced Reynolds number is derived.

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